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IN-LINE AIRCRAFT-ENGINE BEARING LOADS

II - BLADE-BEARING LOADS

By Milton C. Shaw and E. Fred Macks

Aircraft Engine Research Laboratory
Cleveland, Ohio

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ADVANCE RESTRICTED REPORT

IN-LINE AIRCRAFT-ENGINE BEARING LOADS

II - BLADE-BEARING LOADS

By Milton C. Shaw and E. Fred Macks

SUMMARY

A dimensional-analysis method of analyzing bearing loads is applied to the blade bearing of a V-type aircraft engine. This computation permits the determination of blade-bearing operating characteristics at all values of engine speed to 5000 rpm and at all values of indicated mean effective pressure to 500 pounds per square inch.

Optimum combinations of engine speed and indicated mean effective pressure have been found to exist for which the mean and the maximum blade-bearing loads are minima for a given power output. The best dive-throttle setting with regard to blade-bearing load is shown to be one that will produce an indicated mean effective pressure slightly less than that corresponding to the optimum at dive speed. The maximum blade-bearing load occurs when the rubbing velocity is approximately zero.

A production, 12-cylinder, V-type engine is specifically considered to demonstrate the application of the dimensional-analysis method to the generalization of blade-bearing-load computations.

INTRODUCTION

A generalized method of determining aircraft-engine bearing loads at any combination of engine speed and indicated mean effective pressure is developed in reference 1. Detailed descriptions of the theoretical basis of the analysis and the development of the generalized treatment are presented therein; computations of maximum and mean crankpin-bearing loads of a production, 12-cylinder, V-type engine are included to demonstrate the method.

From a series of equations developed by dimensional analysis, the following fundamental equation is derived in reference 1 for calculating the bearing loads of any internal combustion engine in which the indicated mean effective pressure is assumed to be proportional to the manifold pressure:

$$W = L_S{}^2 p \Omega \left(\frac{M_1 N^2}{L_S p}, \frac{M_c N^2}{L_S p}, \frac{D}{L_S}, \frac{L_R}{L_S}, \frac{p_m}{r}, \theta, r, \gamma \right) \tag{1}$$

where

N engine speed, rpm

p indicated mean effective pressure, pounds per square inch

L_S stroke, inches

W bearing load, pounds

M_1 reciprocating mass per crankpin, slugs

M_c rotating mass per crankpin, slugs

D diameter of bore, inches

L_R length of connecting rod, inches

p_m manifold pressure, pounds per square inch absolute

r compression ratio

θ crank angle, degrees

γ angle between cylinder center lines, degrees

Ω some function

Equation (1) simplifies to the following expression for a specific engine

$$W = p\Omega' \left(\frac{N^2}{p}, \theta \right) \tag{2}$$

Equation (2), which is applicable to the principal bearings of both in-line and radial engines, establishes the fact that, if W/p is plotted against N^2/p at constant crank angle, a smooth curve will be obtained. Before equation (2) can be applied and before the computations can be generalized, the blade-bearing loads for a number of representative engine operating conditions must be computed to obtain values of W/p. These computations are made in the usual manner (reference 2) prior to generalization.

The investigation of bearing-load determinations was continued at the Cleveland laboratory of the NACA during the spring of 1944 and analyses of maximum and mean blade-bearing loads of a V-type engine, based on the generalized method, are presented herein.

CONVENTIONAL COMPUTATION OF BLADE-BEARING LOADS

The symbols, the conventions, the engine dimensions, and the method of analysis employed herein are the same as those in reference 1. The power conditions investigated are the same as in reference 1 and are given in table I. (The bearing operating characteristics presented in table I will be considered in the following section.) Throughout this report a crank angle θ of 0° refers to the top-center position of cylinder 1L at the beginning of the expansion stroke.

A schematic diagram of a V-type engine mechanism is shown in figure 1, and the connecting-rod and blade-bearing arrangement is illustrated in figure 2.

The resultant load acting on the blade journal at any particular crank angle is obtained by vector addition of the centrifugal force and the co-linear gas and reciprocating inertia forces. A representative polar diagram of the forces acting upon the blade journal with respect to the engine axis is shown in figure 3. This diagram is for an indicated mean effective pressure of 242 pounds per square inch and an engine speed of 3000 rpm. The individual vectors constituting the resultant blade-bearing load, at a crank angle of 320°, are shown to illustrate the method of vector addition.

A polar diagram with respect to the fork-rod axis is more useful than a diagram with respect to the engine axis for determining the exact load vector acting on the blade journal. These diagrams may be obtained by rotating each resultant vector of figure 3 through an angle ϕ_1 (fig. 1) in the direction of crankshaft rotation. Polar diagrams with respect to the fork-rod axis for each of the six power conditions investigated are given in figures 4 and 5 in terms of crank-angle degrees.

Polar diagrams with respect to the blade-rod axis are of interest inasmuch as they determine the load vectors acting on the blade bearing. Such diagrams are obtained by rotating the individual vectors of figures 4 and 5 through an angle of $120^{\circ} + \delta$ (fig. 1) in the direction of crankshaft rotation. Polar diagrams with respect to the blade-rod axis, for each of the six power conditions investigated, are given in figures 6 and 7 in terms of crank-angle degrees.

APPLICATION OF THE DIMENSIONAL-ANALYSIS METHOD

Generalized Load Charts

Maximum bearing loads. - The resultant blade-bearing forces shown in figures 3 to 7 can be generalized by means of equation (2). When W/p is plotted against N^2/p for each of the six power conditions at constant values of crank angle (fig. 8), the maximum values of W/p are found to occur at crank angles of approximately 320° or 660°. Additional points were computed for these crank angles in order to extend the curves in figure 8 beyond the region covered by the six power conditions. The solid portion of each curve corresponds very closely to the maximum value of W/p over the particular range of N^2/p concerned. All the plots of W/p against N^2/p are portions of hyperbolic-type curves. The solid portions of both curves of interest lie sufficiently far from their respective vertices to be considered linear.

A convenient chart for determining maximum blade-bearing loads, obtained from the curves of figure 8, is presented in figure 9. The line OA represents an optimum combination of speed and indicated mean effective pressure corresponding to the lowest possible value of the maximum blade-bearing load at a given power level. It can be seen from figure 9 that this optimum combination falls in an impractical operating region. Constant indicated-horsepower curves have been included for convenience.

Mean bearing loads. - The mean load acting on the blade bearing is determined by plotting load against crank angle on coordinate graph paper; the average height of this curve is obtained by use of a polar planimeter. The results of the dimensional treatment were also utilized to generalize the mean-load analysis. In figure 10, $\overline{W}/p$ is plotted against N^2/p, where $\overline{W}$ is the mean blade-bearing load.

A useful chart for determining mean blade-bearing loads is obtained from figure 10 by plotting engine speed against indicated mean effective pressure for constant values of mean load. Such a family of curves is presented in figure 11. The optimum-maximum-load curves shown for four compression ratios were included to permit comparison with each other as well as with the mean-load curves and will be discussed under Effect of Engine Dimensions upon Blade-Bearing Loads. Constant indicated-horsepower curves have been included for convenience.

Optimum combinations of indicated mean effective pressure and engine speed. - Families of constant indicated-horsepower curves (fig. 12) for maximum and mean blade-bearing loads are obtained from figures 9 and 11. (Points beyond the range of these charts were obtained from figs. 8 and 10.) The loci of the optimum combinations of engine speed and indicated mean effective pressure for the maximum and mean blade-bearing loads are represented by the curves CC and DD, respectively.

A closed throttle setting in a dive is desirable with regard to the mean blade-bearing load, as shown in figure 11. The rate of increase of maximum blade-bearing load with indicated mean effective pressure at constant speed is great for points above line OA of figure 9 but is actually negative for points below OA. The indicated mean effective pressure will therefore affect the maximum bearing load very little if the point representing the dive speed and the indicated mean effective pressure lies below line OA. Part-throttle operation corresponding to a point close to, but below, line OA should minimize any tendency toward oil pumping in a dive.

Rubbing factor. - The "rubbing factor," although considered a poor criterion for evaluating the severity of bearing operating conditions, will be determined. The customary method of determining this factor, multiplying the rubbing speed of a journal by the mean unit-bearing load, must be extended in order that the rubbing factor of the blade bearing may be obtained. The relative motion between the blade bearing and the blade journal is shown in the appendix to be an approximate sinusoidal oscillation. Inasmuch as the rubbing factor is generally considered to be proportional to the heat generated in the bearing, it is herein considered as the product of the mean relative rubbing velocity and the mean unit bearing load. The mean rubbing velocity $\overline{V}$ (in ft/sec) is determined from equation (A12) (in the appendix):

$$\overline{V} = 0.00308\ N$$

The rubbing factor RF for the blade bearing is therefore

$$RF = 0.000897 \, N\bar{W} \tag{2}$$

where the effective bearing area is 3.44 square inches and the rubbing factor is given in units of (ft lb)/(sq in.)(sec).

Verification of the Generalized Load Charts

The generalized analysis was checked by constructing a polar diagram for an extreme combination of speed and indicated mean effective pressure (3600 rpm and 182 lb/sq in.). The resulting polar diagram is shown in figure 13. The maximum load from figure 13 is 12,450 pounds at a crank angle of 660°. The corresponding maximum load from figure 9 is 12,250 pounds also at a crank angle of 660°. The close agreement of values is considered as ample verification of the accuracy of the computations as well as of the generalized treatment.

Effect of Engine Dimensions upon Blade-Bearing Loads

The blade-load charts presented are directly applicable only to an engine having the same dimensions as the V-type engine herein considered. As pointed out in reference 1, attempts to make bearing-load diagrams applicable to any in-line engine have not been entirely successful inasmuch as no simple method has been found by which a change in the magnitude of the reciprocating and the rotating weights may be taken into consideration when dimensional analysis is employed. Changes in load charts brought about by differences in the connecting-rod length and the compression ratio have, however, been determined. In general, the engine dimensions have the same effect upon blade-bearing loads as upon crankpin-bearing loads, which are described and explained in detail in reference 1. For convenience of the reader these effects are given herewith as they apply to the blade bearing:

1. Changes in the ratio of connecting-rod length to crank throw within the range from 3 to 4 have no measurable effect upon the accuracy of the values of blade-bearing loads given in the polar diagrams.

2. The compression ratio affects the shape of the indicator diagram and therefore affects the gas force developed in the engine cylinder, particularly during that portion of the expansion stroke when the piston is near the top-center position.

3. The compression ratio considerably influences the resultant maximum blade-bearing loads in the crank-angle region of 320°. (See figs. 9 and 14.)

4. The compression ratio will influence the mean blade-bearing load very little because the compression ratio significantly affects the gas force only during a small portion of the cycle and part of this effect is compensatory.

5. Location of the curves OA for optimum combinations of speed and indicated mean effective pressure changes with compression ratio. (See fig. 11.)

DISCUSSION

Representative values of blade-bearing operating characteristics are given in table I for six power conditions. It can be seen that the maximum load increases very rapidly with indicated mean effective pressure when the engine speed is held constant. The maximum load occurs at a crank angle of 320° for practical operating conditions. The maximum unit blade-bearing load is large, especially at high indicated mean effective pressures and low engine speeds. The mean load increases with either an increase in indicated mean effective pressure or an increase in engine speed. The rubbing factor for this bearing is unusually low.

The relative motion between the blade bearing and the journal, as shown in the appendix, varies sinusoidally with an amplitude of approximately $1\frac{1}{8}$ inches. This type of motion together with an unusually small length-diameter ratio of the blade bearing, discourages circumferential oil flow; hence a large number of oil holes is required. (The blade journal of the V-type engine herein considered is fitted with eight equally spaced holes of 7/64-in. diameter.)

Figures 4 to 7 indicate that the maximum loads occur near the crank-angle values at which the rubbing velocity is 0, which is undesirable because the bearing must start from rest under very high load. The possibility of altering the firing order to prevent the maximum blade-bearing load from occurring at a rubbing velocity of 0 was investigated. It was found that, for all practical firing orders of a V-type engine, the right-block piston must be at a crank angle of either 60° or 420° when the left-block piston is at 0°. The 0 value of velocity occurs at the maximum load for each of these phase relations and, therefore, for all feasible firing orders.

For oscillatory motion, load is best carried by a rolling-contact bearing. If difficulty is experienced with the blade bearing at high values of indicated mean effective pressure, the use of a needle blade bearing might be a likely solution. Inasmuch as the load at the bearing parting lines is always relatively small, no difficulty should be experienced because of discontinuity of the bearing surface at the parting line.

CONCLUSIONS

From a series of computations using the dimensional-analysis method of analyzing the blade-bearing loads of a V-type engine, the following conclusions were drawn.

For V-type engines:

1. Optimum combinations of engine speed and indicated mean effective pressure exist for which the mean and maximum blade-bearing loads are minima for a given power output.

2. At a given power level the optimum maximum blade-bearing load varies directly with the compression ratio.

3. The ratio of connecting-rod length to crank throw does not appreciably influence the mean or the maximum blade-bearing load.

For the production, V-type engine herein considered:

1. The combinations of engine speed and indicated mean effective pressure corresponding to optimum values of both mean and maximum blade-bearing loads lie in an impractical operating region.

2. The maximum blade-bearing load occurs in the crank-angle regions of 320° or 660° depending upon the relative values of engine speed and indicated mean effective pressure employed.

3. The maximum blade-bearing load occurs when the rubbing velocity is approximately 0.

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APPENDIX - RELATIVE MOTION BETWEEN BLADE JOURNAL AND BLADE BEARING

The following derivation can be used to obtain an expression for determining the mean rubbing velocity of the blade journal relative to the blade bearing. From figure 1:

$$\delta = \frac{\pi}{3} + \phi_2 - \phi_1 \tag{A1}$$

$$R \sin \theta = L_R \sin \phi_1 \tag{A2}$$

$$R \sin\left(\theta + \frac{\pi}{3}\right) = L_R \sin \phi_2 \tag{A3}$$

where all angles are measured in radians. From equations (A1), (A2), and (A3)

$$\delta = \frac{\pi}{3} + \sin^{-1}\left[\frac{R}{L_R} \sin\left(\theta + \frac{\pi}{3}\right)\right] - \sin^{-1}\left(\frac{R}{L_R} \sin \theta\right) \tag{A4}$$

For the V-type engine herein considered, R/L_R is 0.30. Therefore

$$\delta = \frac{\pi}{3} + \sin^{-1}\left[0.30 \sin\left(\theta + \frac{\pi}{3}\right)\right] - \sin^{-1}(0.30 \sin \theta) \tag{A5}$$

Inasmuch as the blade bearing of the V-type engine herein considered is 3.688 inches in diameter, the distance between the fork-rod axis and the blade-rod axis measured along the bearing periphery is

$$s = \frac{3.688}{24} \delta \text{ ft} \tag{A6}$$

From equations (A5) and (A6)

$$s = 0.1536\left\{\frac{\pi}{3} + \sin^{-1}\left[0.30 \sin\left(\theta + \frac{\pi}{3}\right)\right] - \sin^{-1}(0.30 \sin \theta)\right\} \tag{A7}$$

Equation (A7) may be approximated by the following simplified expression with a maximum error of 0.5 percent:

$$s = 0.1617 + 0.0467 \sin\left(\theta + \frac{2\pi}{3}\right) \tag{A8}$$

The relative velocity V between the blade bearing and blade journal may be obtained by differentiating expression (A7) with respect to time:

$$V = 0.00482\ N \left\{ \frac{\cos\left(\theta + \frac{\pi}{3}\right)}{\sqrt{1 - \left[0.30 \sin\left(\theta + \frac{\pi}{3}\right)\right]^2}} - \frac{\cos\ \theta}{\sqrt{1 - (0.30 \sin\ \theta)^2}} \right\} \quad \text{(A9)}$$

where N is engine speed, rpm.

Equation (A9) may be approximated with a maximum error of 4 percent in the following manner:

$$V = 0.00482\ N \cos\left(\theta + 2\frac{\pi}{3}\right) \quad \text{(A10)}$$

The time-weighted mean velocity $\overline{V}$ is given by

$$\overline{V} = \frac{\int_0^{\frac{30}{N}} V dt}{\int_0^{\frac{30}{N}} dt} \quad \text{(A11)}$$

If zero time corresponds to the crank-angle value for which the velocity is $0 \left(\frac{5\pi}{6} \text{ radians}\right)$, then

$$\overline{V} = \frac{s \Big|_{\frac{5\pi}{6}}^{\frac{11\pi}{6}}}{\frac{30}{N}} = 0.00308\ N \text{ ft/sec} \quad \text{(A12)}$$

1. Shaw, Milton C., and Macks, E. Fred: In-Line Aircraft-Engine Bearing Loads. I - Crankpin-Bearing Loads. NACA ARR No. E5H10a, 1945.

2. Prescott, Ford L., and Poole, Roy B.: Bearing-Load Analysis and Permissible Loads as Affected by Lubrication in Aricraft Engines. SAE Jour., vol. XXIX, no. 4, Oct. 1931, pp. 296-315; II. SAE Jour., vol. XXIX, no. 5, Nov. 1931, pp. 379-389; discussion, pp. 389-390.

TABLE I - REPRESENTATIVE VALUES OF BLADE-BEARING OPERATING CHARACTERISTICS

FOR A PRODUCTION, 12-CYLINDER, V-TYPE ENGINE

Power condition	Engine speed, N (rpm)	imep, p (lb/sq in.)	ihp	Maximum bearing load[a], W (lb)	Maximum unit bearing load[b] (lb/sq in.)	Location of maximum bearing load (crank-angle deg)	Mean bearing load[a], $\overline{W}$ (lb)	Mean unit bearing load[b] (lb/sq in.)	Rubbing factor, RF (ft lb)/(sq in.)(sec)
1	3000	182	1170	10,070	2920	320	5380	1560	14,400
2	3000	242	1570	16,160	4700	320	5960	1730	16,000
3	3000	303	1960	22,260	6480	320	6580	1920	17,700
4	3000	363	2350	28,340	8230	320	7100	2060	19,000
5	3300	242	1720	14,640	4250	320	6630	1940	19,800
6	3600	242	1880	12,500	3640	320	7560	2200	24,400

[a]The bearing-load data were taken from figures 9 and 11 and deviate slightly from the values shown on the polar diagrams (figs. 3 to 7).
[b]The projected area of the blade bearing is taken as 3.44 sq in.

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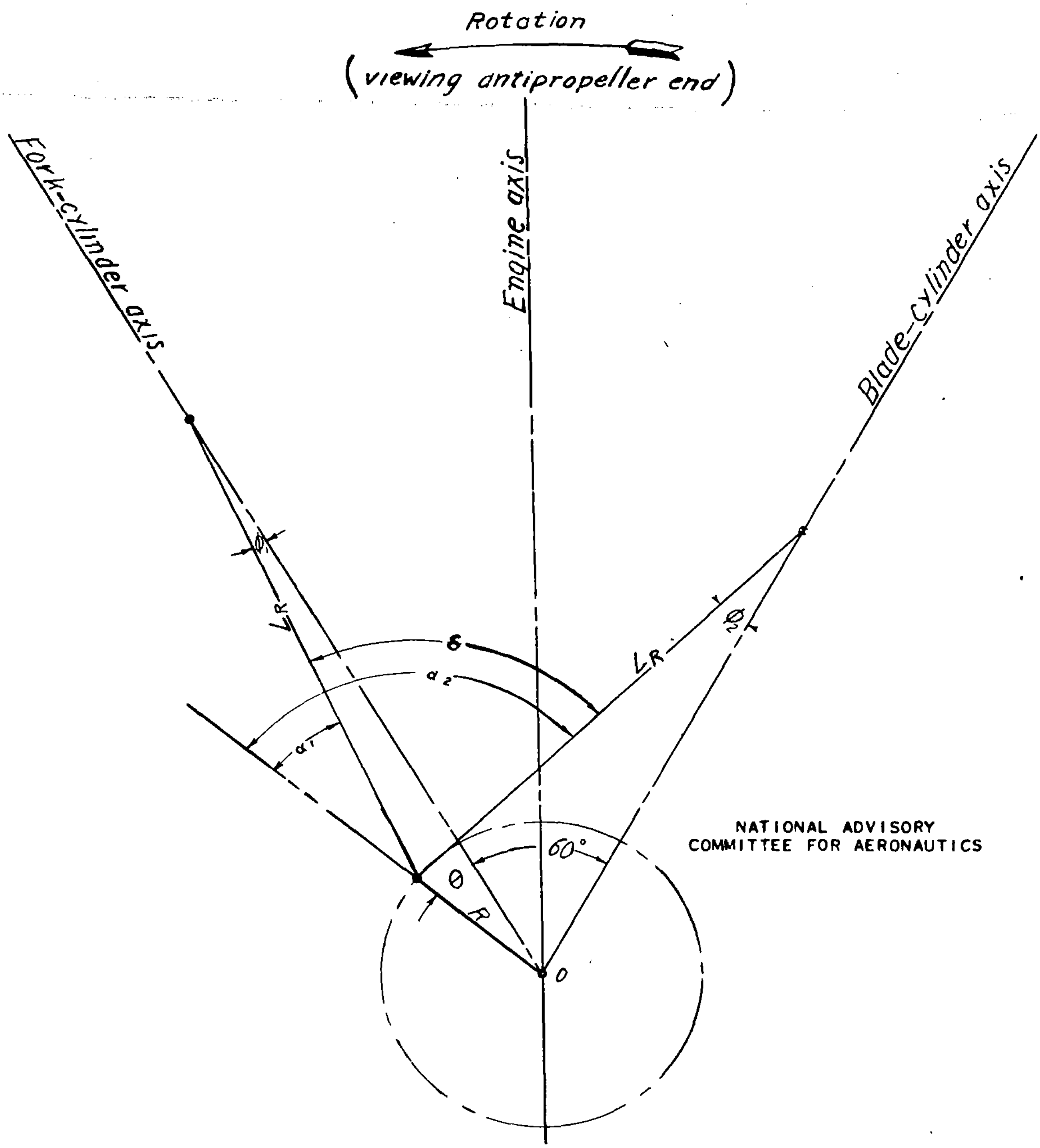

Figure 1. - Schematic diagram of the mechanism of a 12 cylinder V-type engine.

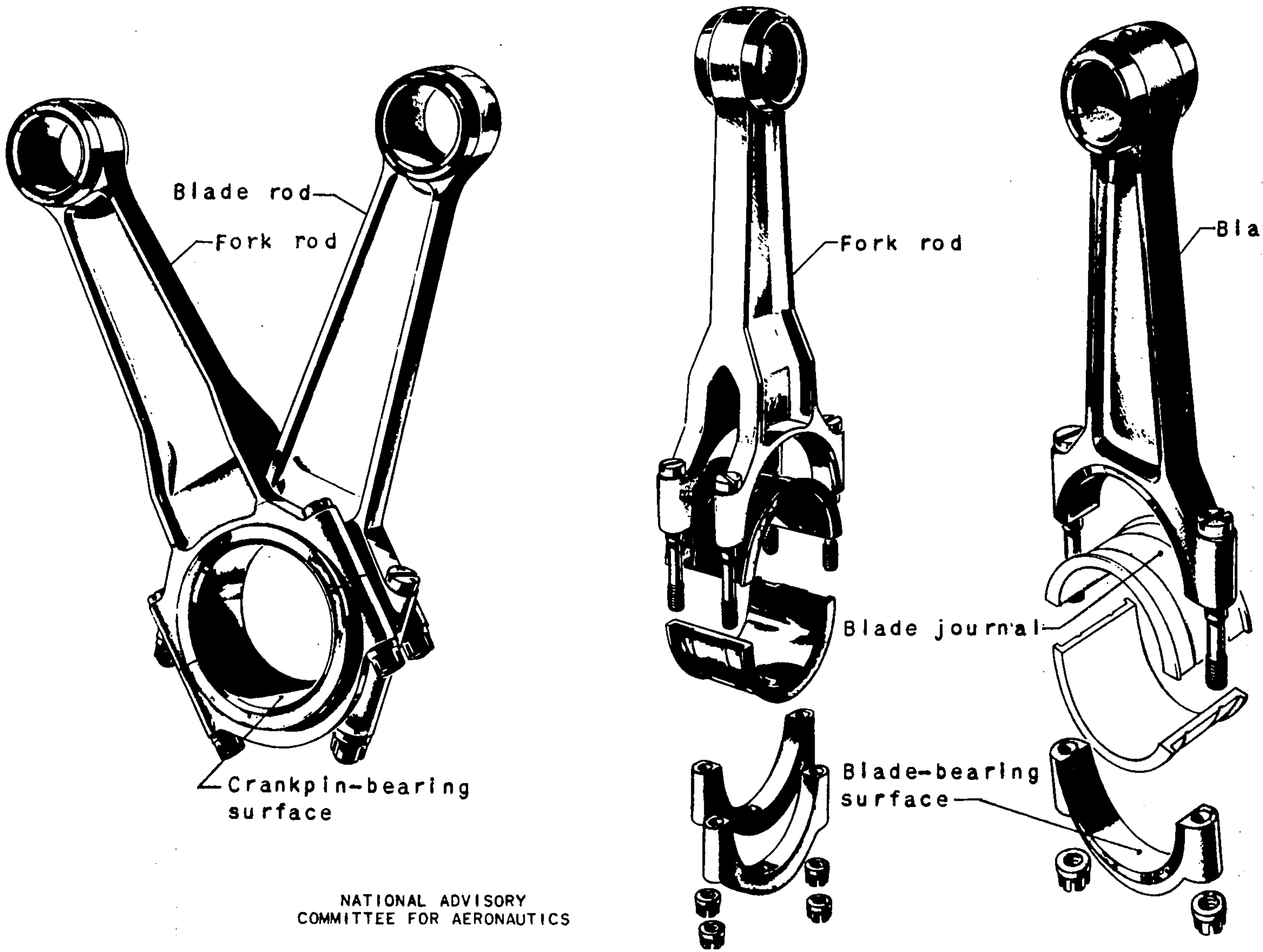

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Figure 2. - Connecting-rod and blade-bearing arrangement for a V-type engine.

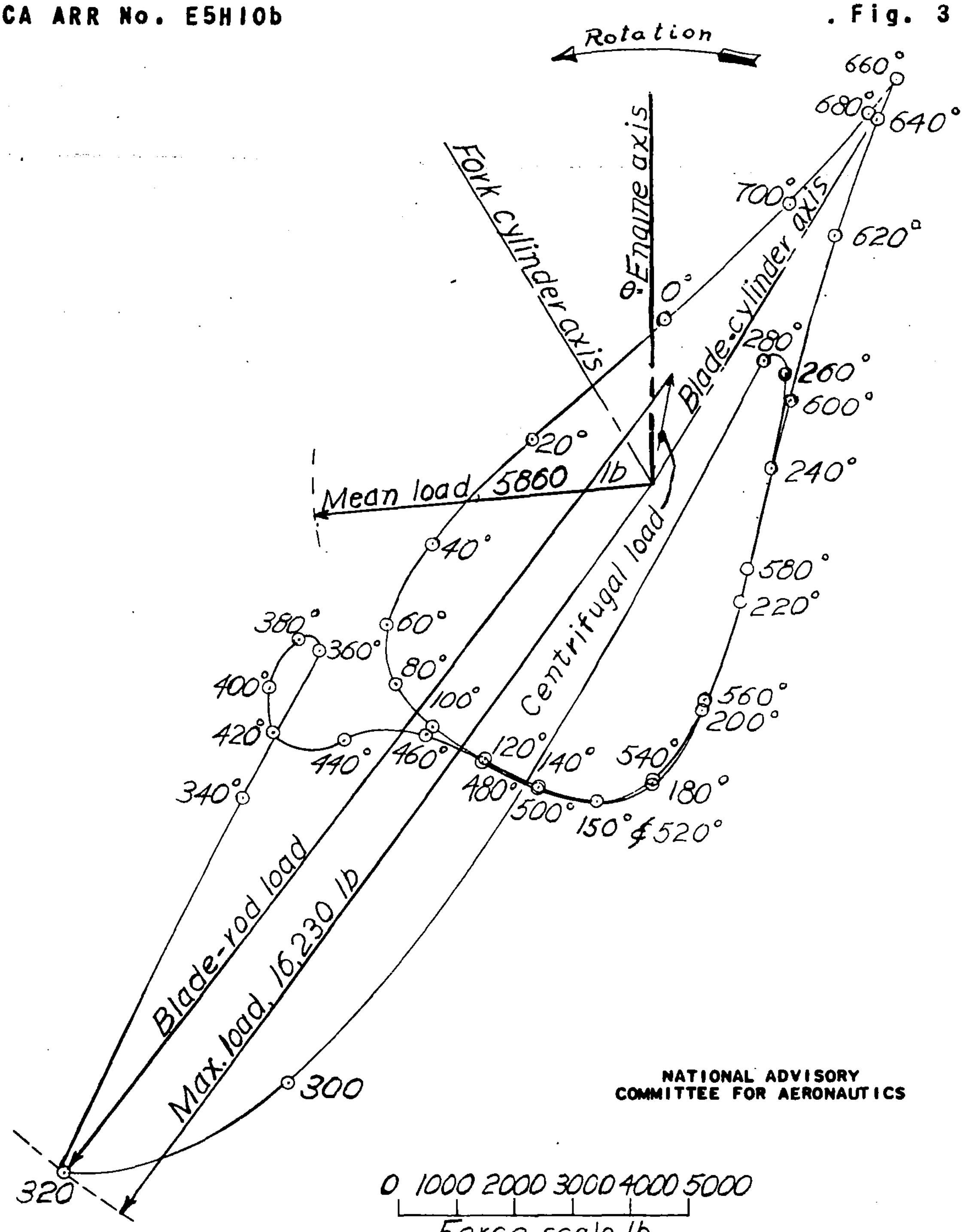
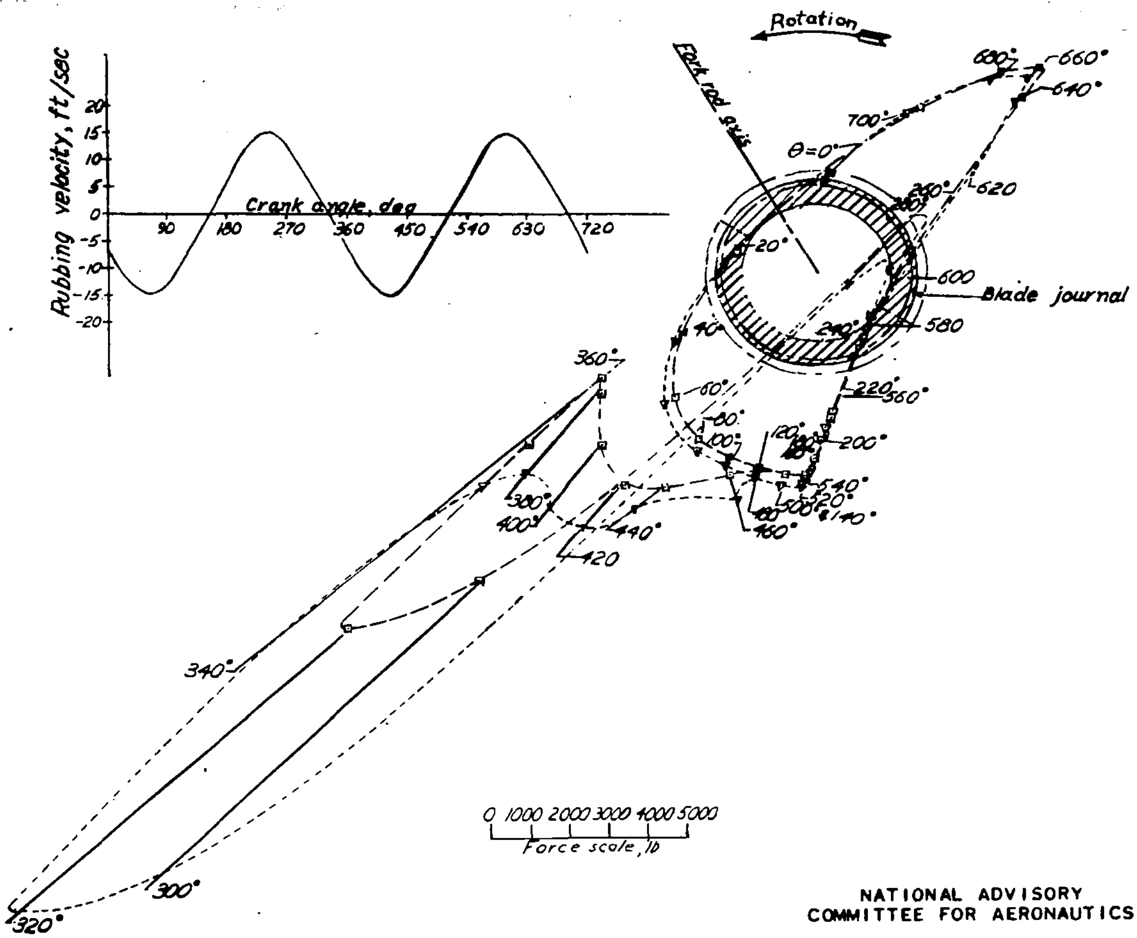


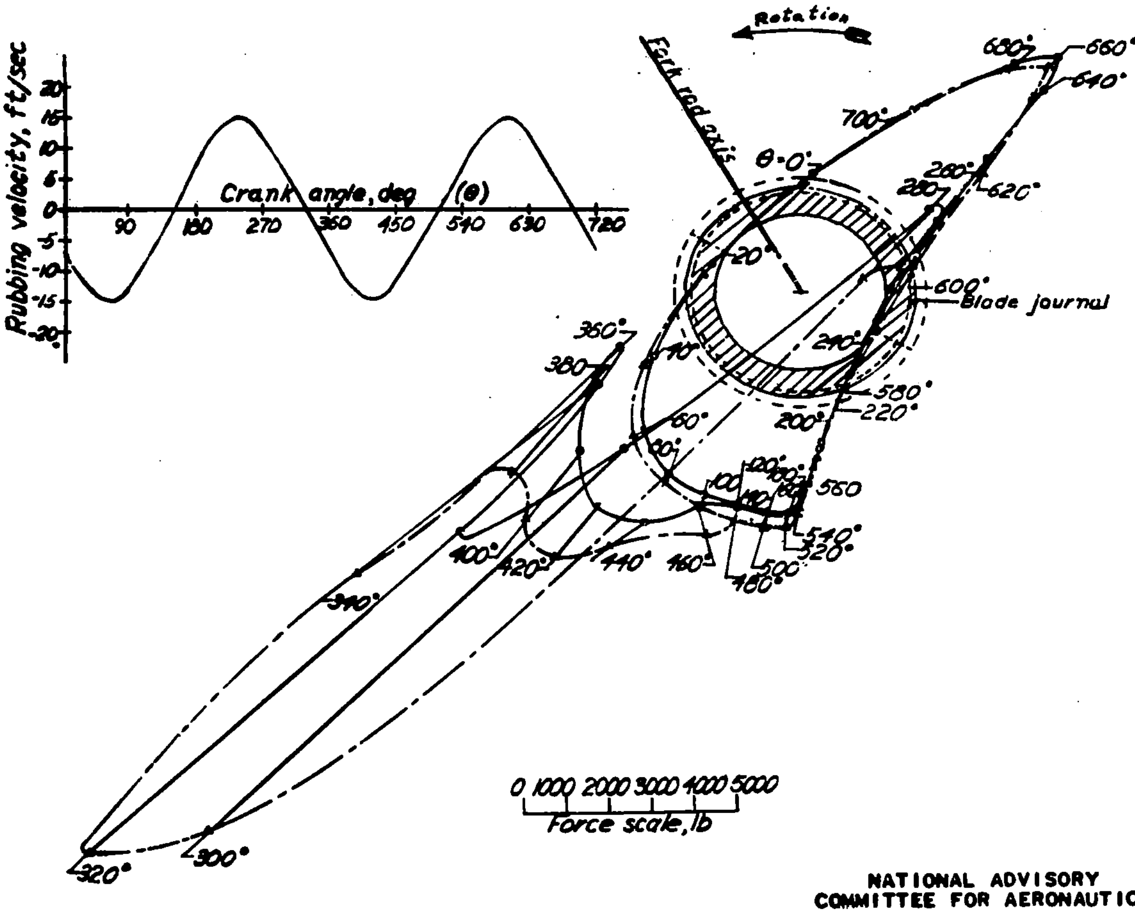
Figure 3. - Polar diagram showing the magnitude of the resultant force on the blade journal of a V-type engine and its direction with respect to the engine axis. Engine speed, 3000 rpm; indicated mean effective pressure, 242 pounds per square inch.



imep, 242 lb/sq in.; maximum force, 16,230 lb at 320°; mean force, 5860 lb
imep, 363 lb/sq in.; maximum force, 28,330 lb at 320°; mean force, 7210 lb

(a) Indicated mean effective pressures, 242 and 363 pounds per square inch.

Figure 4. - Polar diagrams showing the magnitude of the resultant force on the blade journal of a V-type engine and its direction with respect to the fork-rod axis at an engine speed of 3000 rpm.



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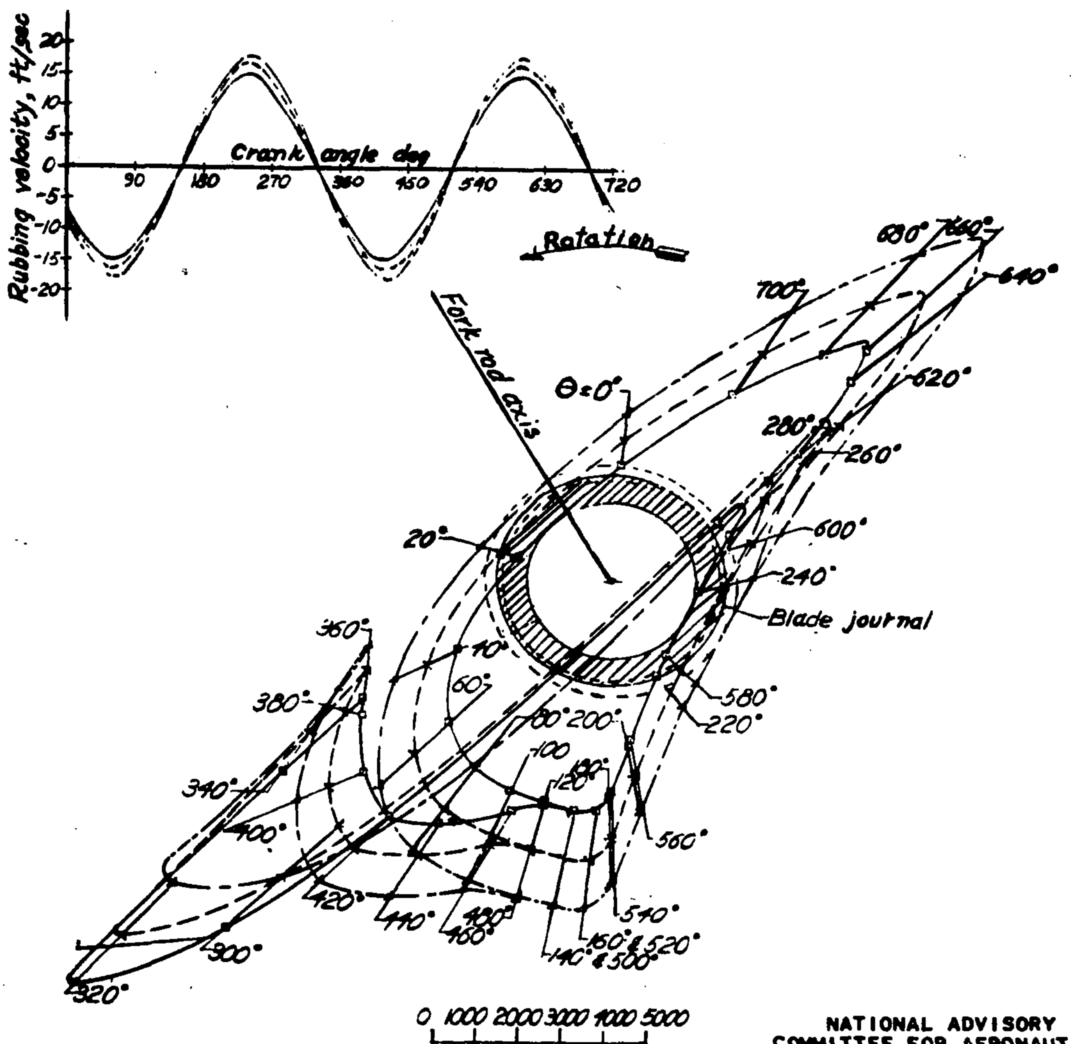
imep, 182 lb/sq in.; maximum force, 10,130 lb at 320°; mean force, 5150 lb
imep, 303 lb/sq in.; maximum force, 22,300 lb at 320°; mean force, 6590 lb

(b) Indicated mean effective pressures, 182 and 303 pounds per square inch.

Figure 4. - Concluded.



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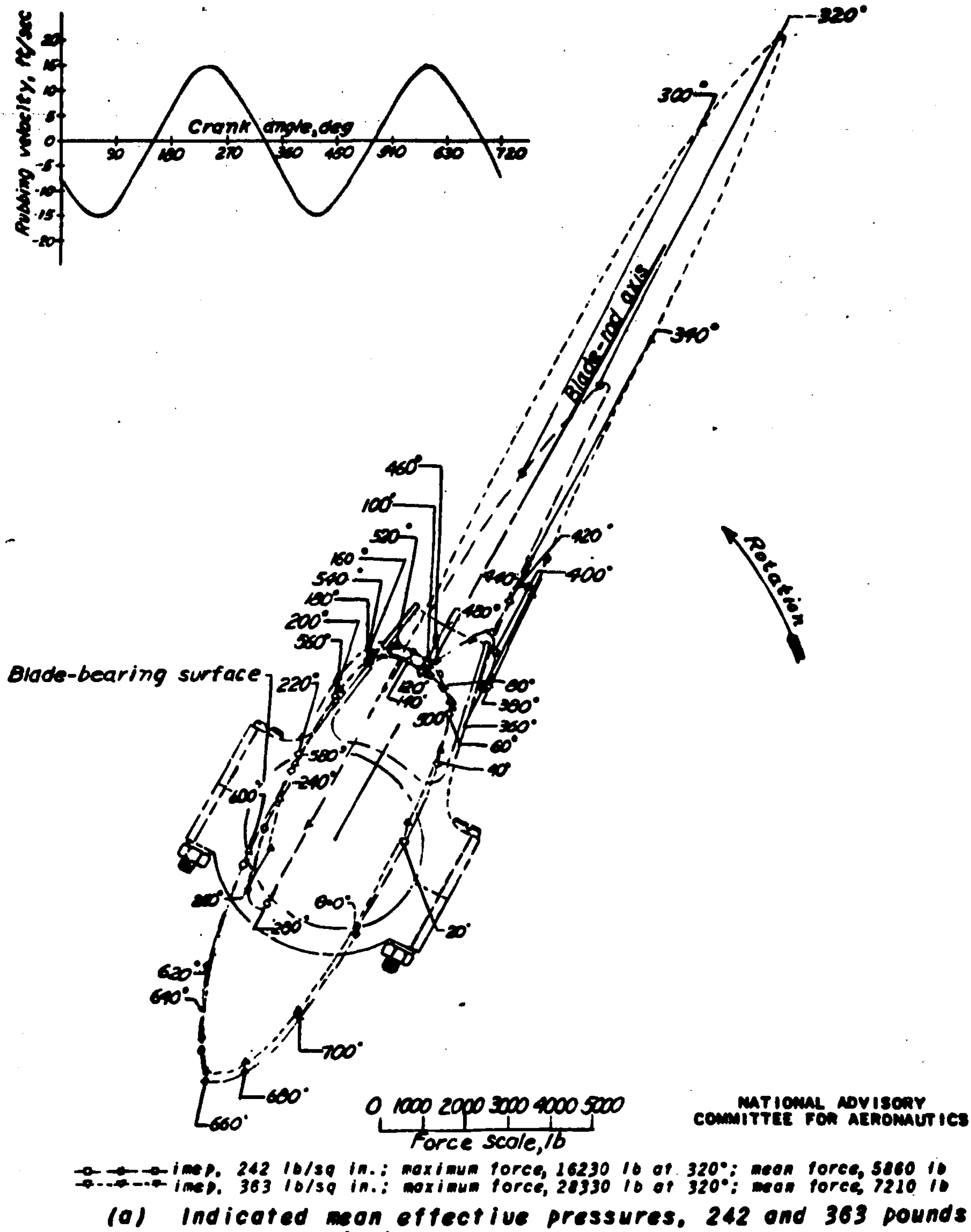
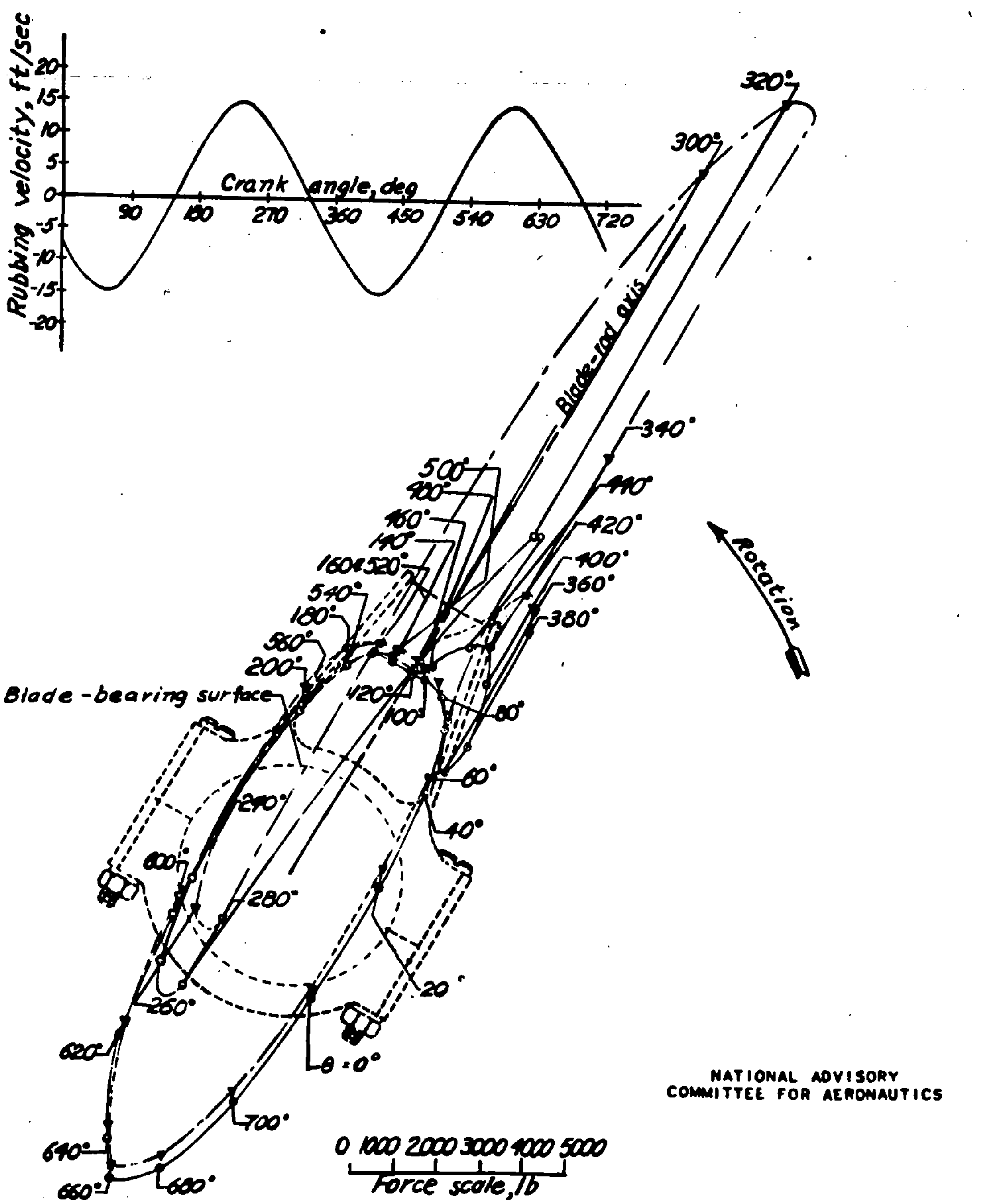
3000 rpm; maximum force, 18,230 lb at 320°; mean force, 5860 lb
3300 rpm; maximum force, 14,580 lb at 320°; mean force, 6720 lb
3600 rpm; maximum force, 12,800 lb at 320°; mean force, 7530 lb

Figure 5. - Polar diagrams showing the magnitude of the resultant force on the blade journal of a V-type engine and its direction with respect to the fork-rod axis at an indicated mean effective pressure of 242 pounds per square inch.



(a) Indicated mean effective pressures, 242 and 363 pounds per square inch.

Figure 6. - Polar diagrams showing the magnitude of the resultant force on the blade bearing of a V-type engine and its direction with respect to the blade-rod axis at an engine speed of 3000 rpm.



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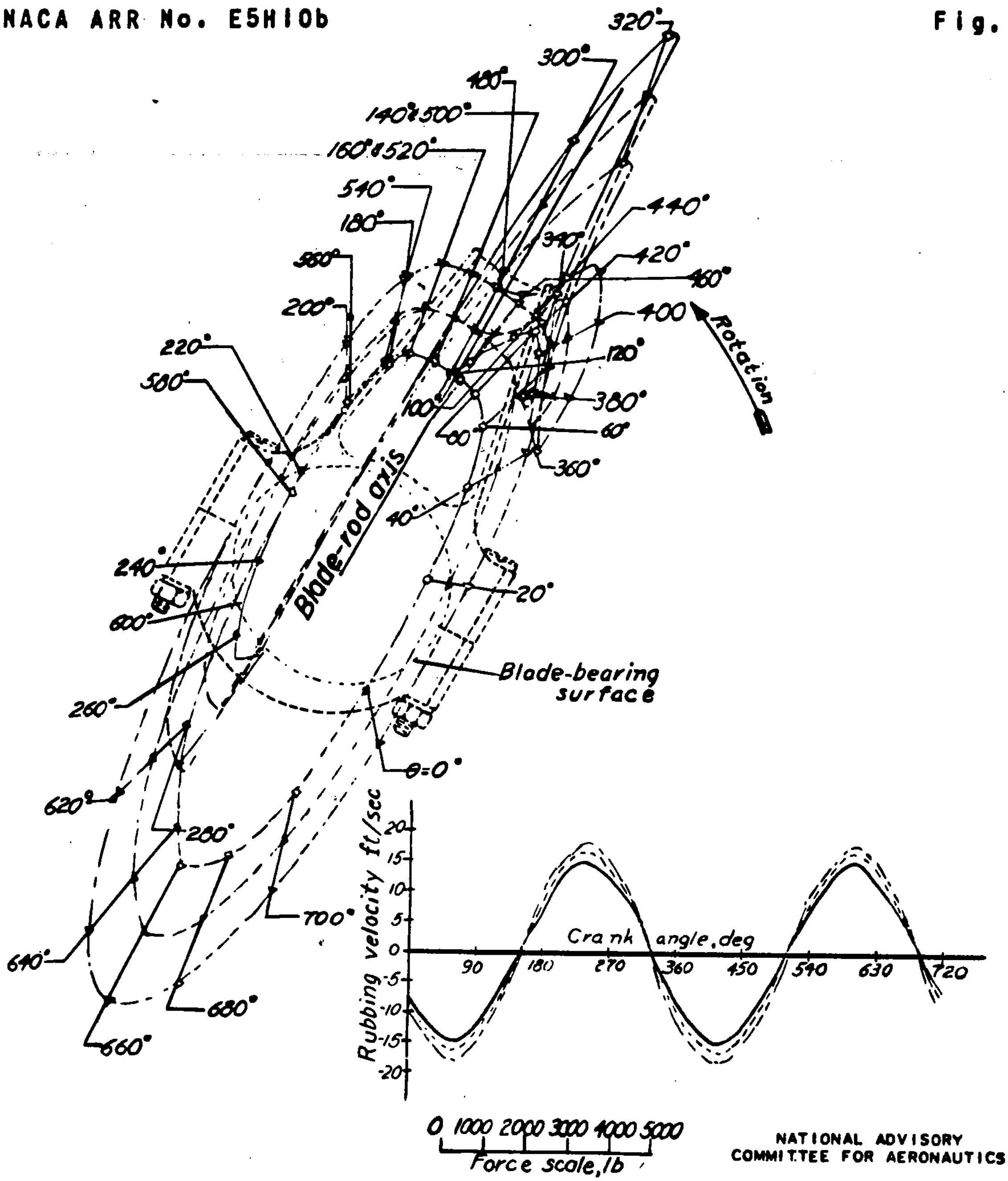
—o—o—o— imep, 182 lb/sq in.; maximum force, 10,130 lb at 320°; mean force, 5150 lb
—▲—▲—▲ imep, 303 lb/sq in.; maximum force, 22,300 lb at 320°; mean force, 6590 lb

(b) Indicated mean effective pressures, 182 and 303 pounds per square inch.

Figure 6. - Concluded.



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3000 rpm; maximum force, 16,230 lb at 320°; mean force, 5860 lb
3300 rpm; maximum force, 14,580 lb at 320°; mean force, 6720 lb
3600 rpm; maximum force, 12,800 lb at 320°; mean force, 7530 lb

Figure 7. - Polar diagrams showing the magnitude of the resultant force on the blade bearing of a V-type engine and its direction with respect to the blade-rod axis at an indicated mean effective pressure of 242 pounds per square inch.

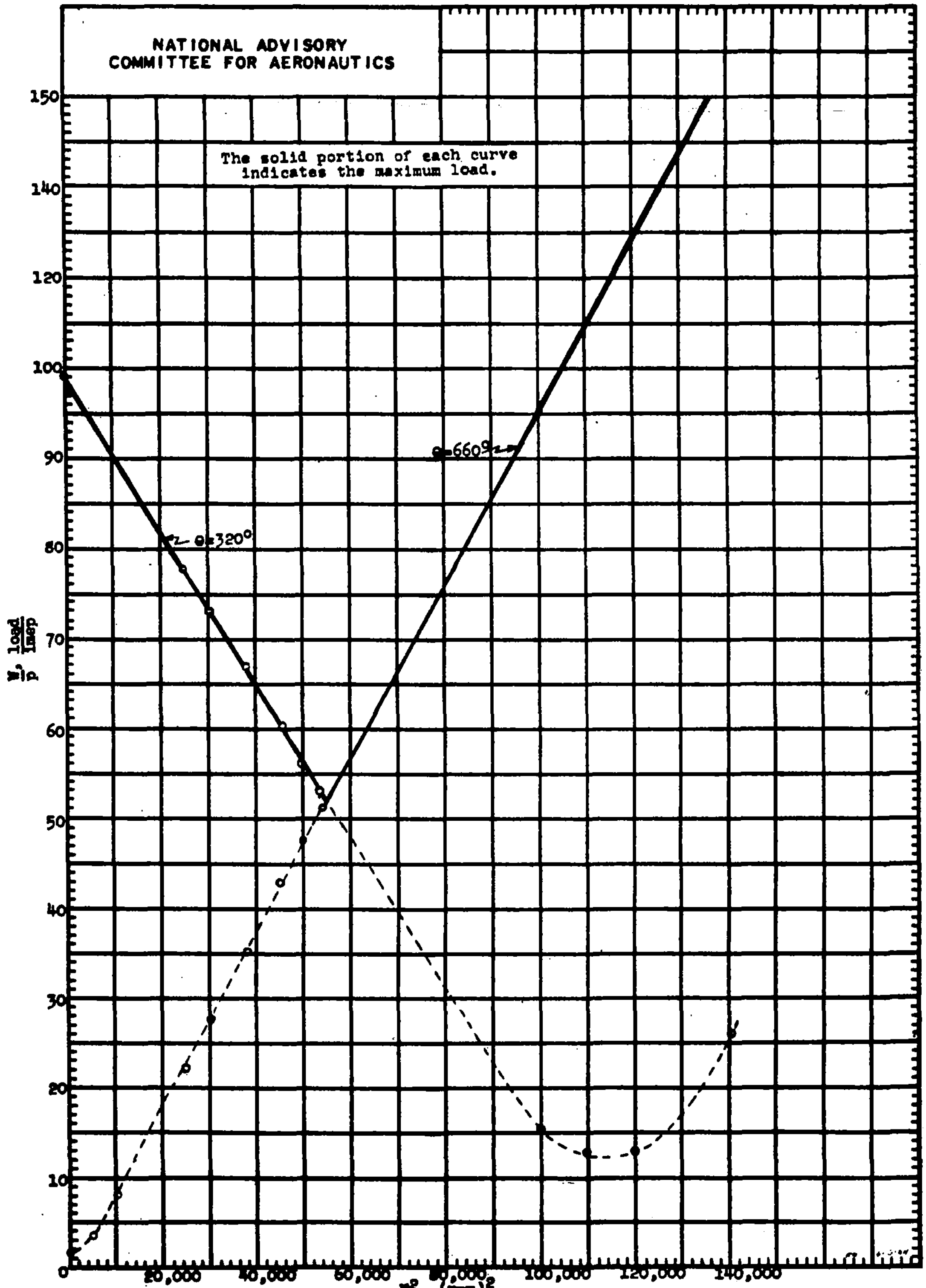

Figure 8. - Representative plot showing the variations of W/p with N^2/p at crank angles of 320° and 660° for a V-type engine.

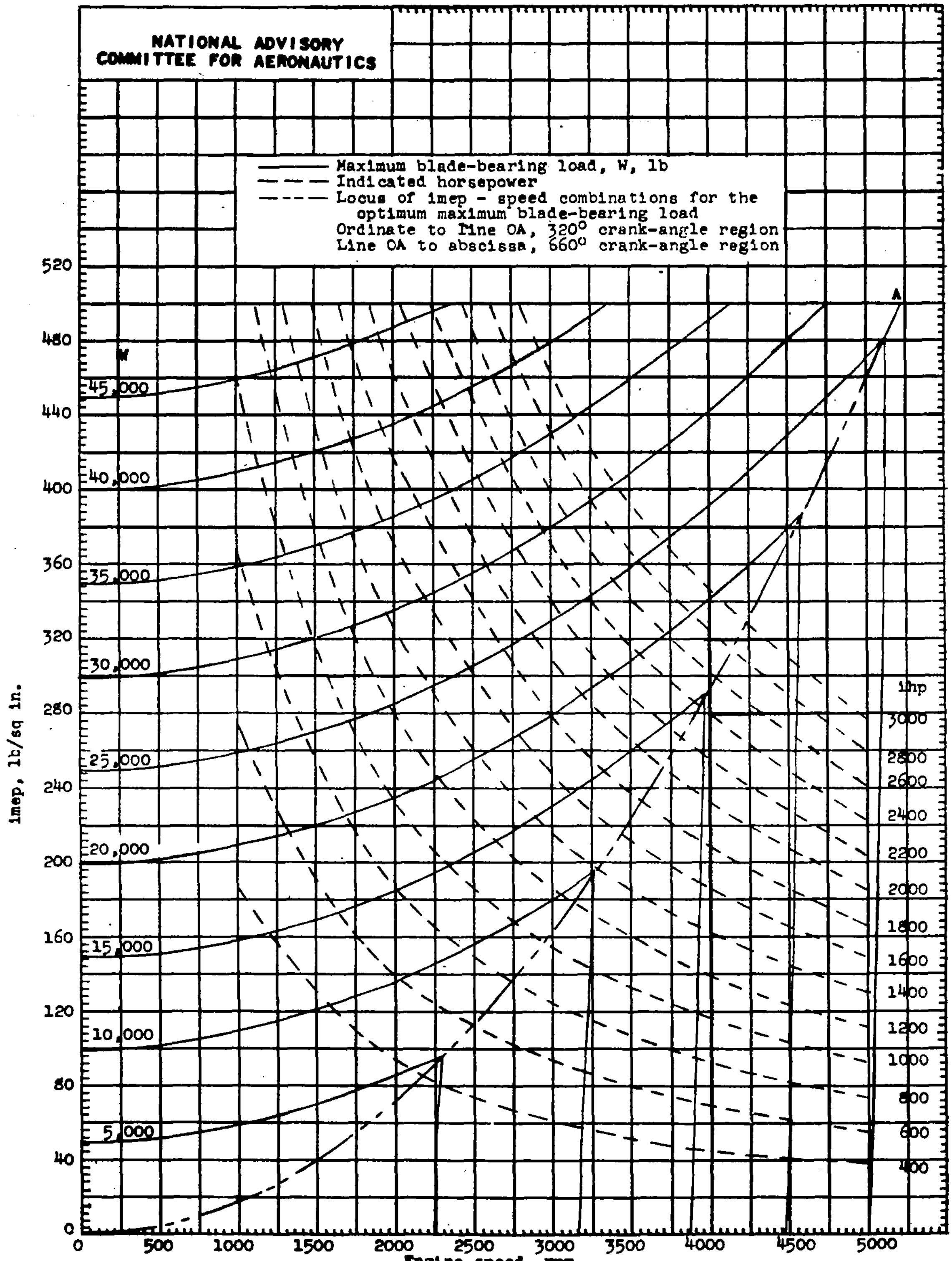

Figure 9. - Maximum load on blade bearing of a production V-type engine for all values of indicated mean effective pressure and engine speed at a compression ratio of 6.65. (Constant maximum-load curves.) Effective bearing area, 3.44 square inches.

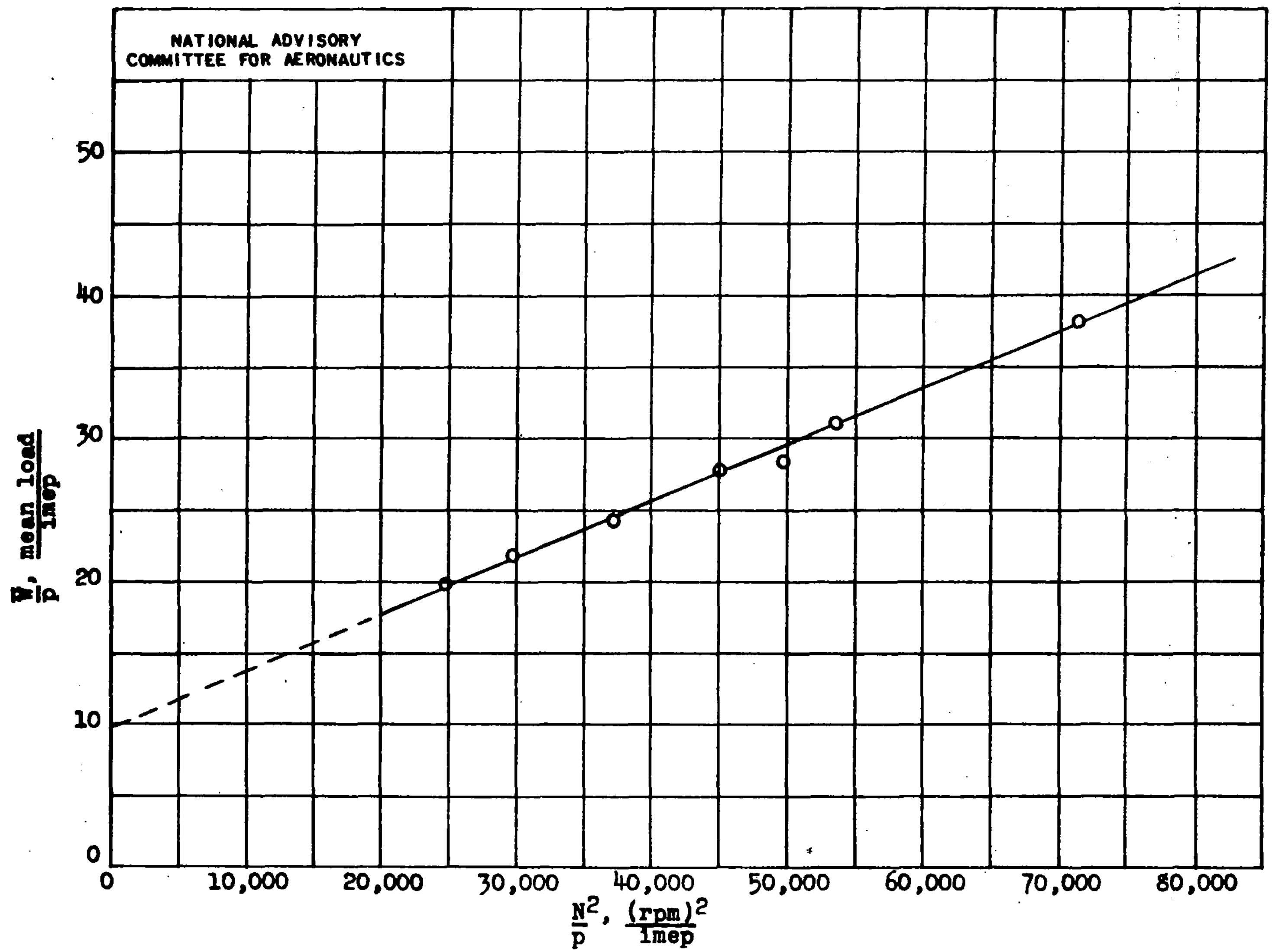

Figure 10. - Variation of $\bar{W}/p$ with N^2/p for blade bearing of a V-type engine.

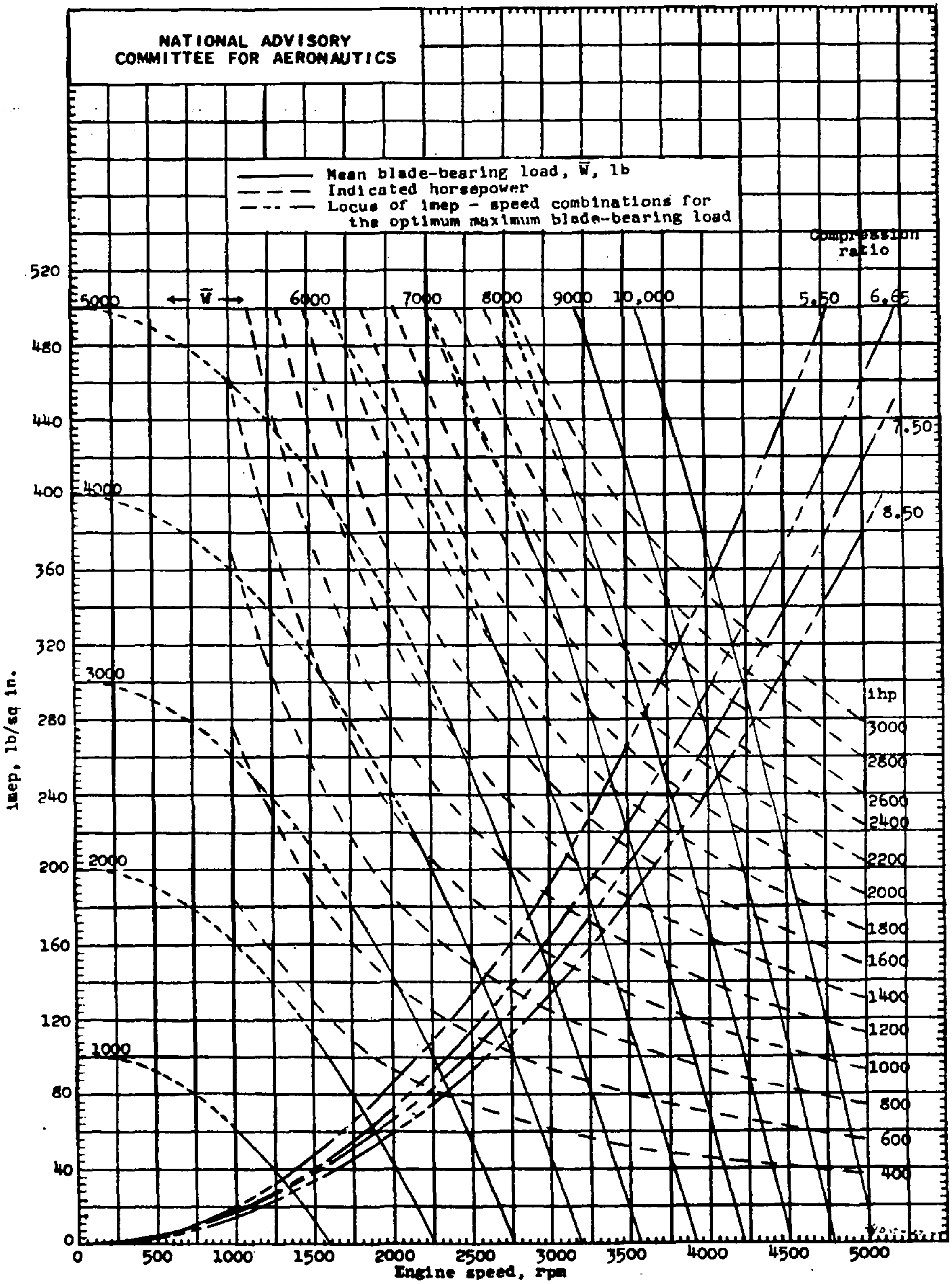

Figure 11. - Mean load on blade bearing of a production V-type engine for all values of indicated mean effective pressure and engine speed. (Constant mean-load curves.) Effective bearing area, 3.44 square inches.

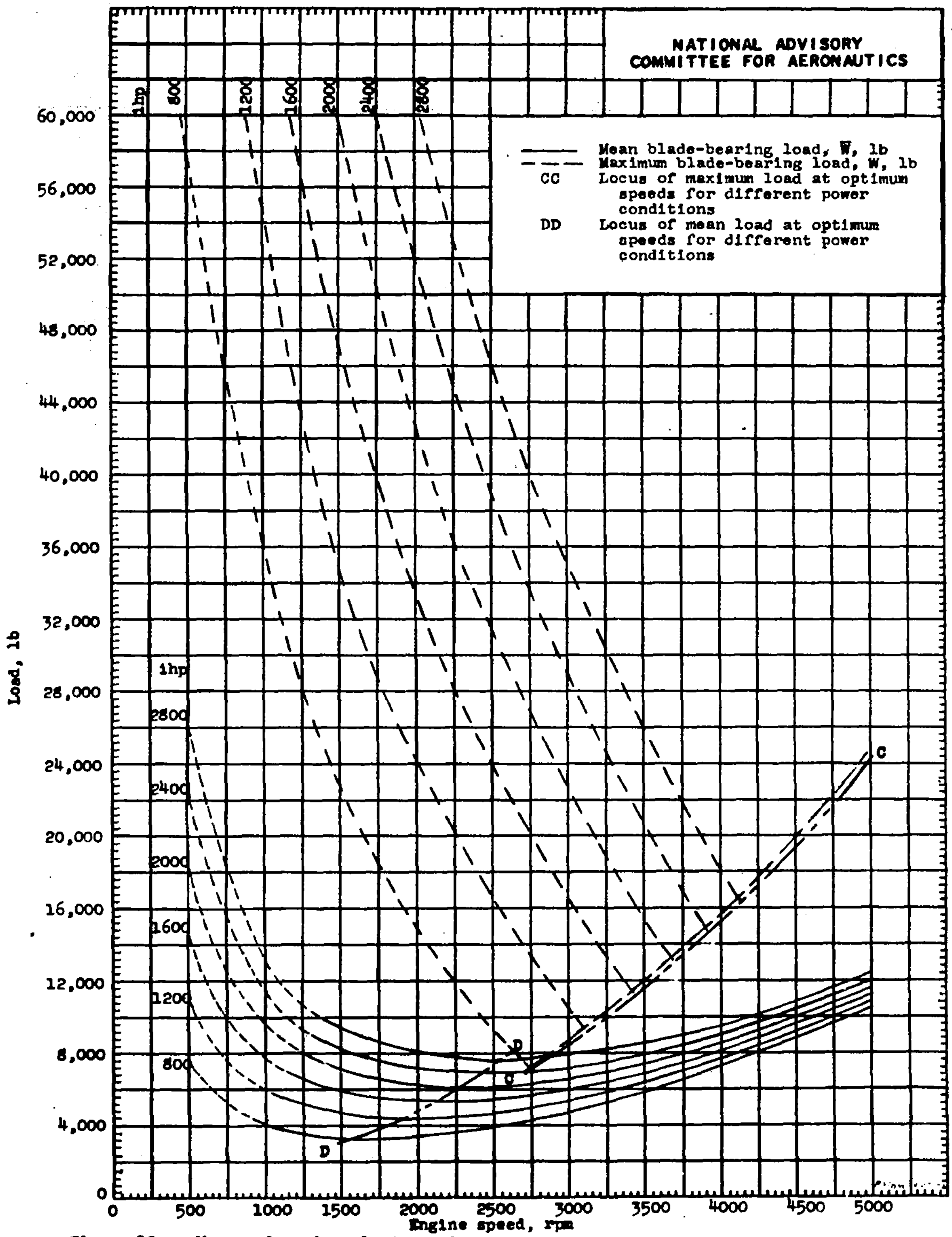

Figure 12. - Mean and maximum load on blade bearing of a V-type engine for all values of indicated horsepower and engine speed at a compression ratio of 6.65. (Constant indicated-horsepower curves.) Effective bearing area, 3.44 square inches.

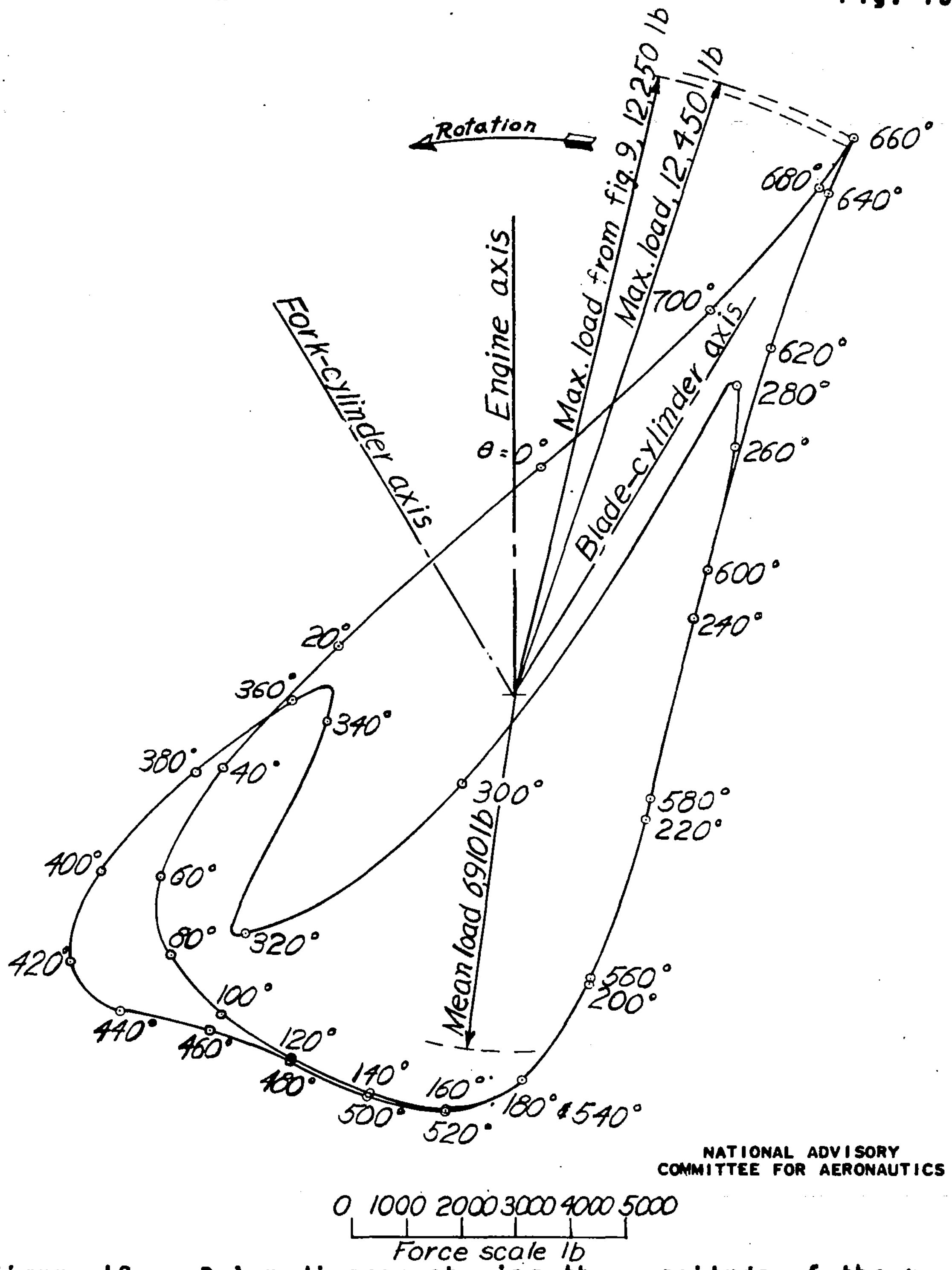

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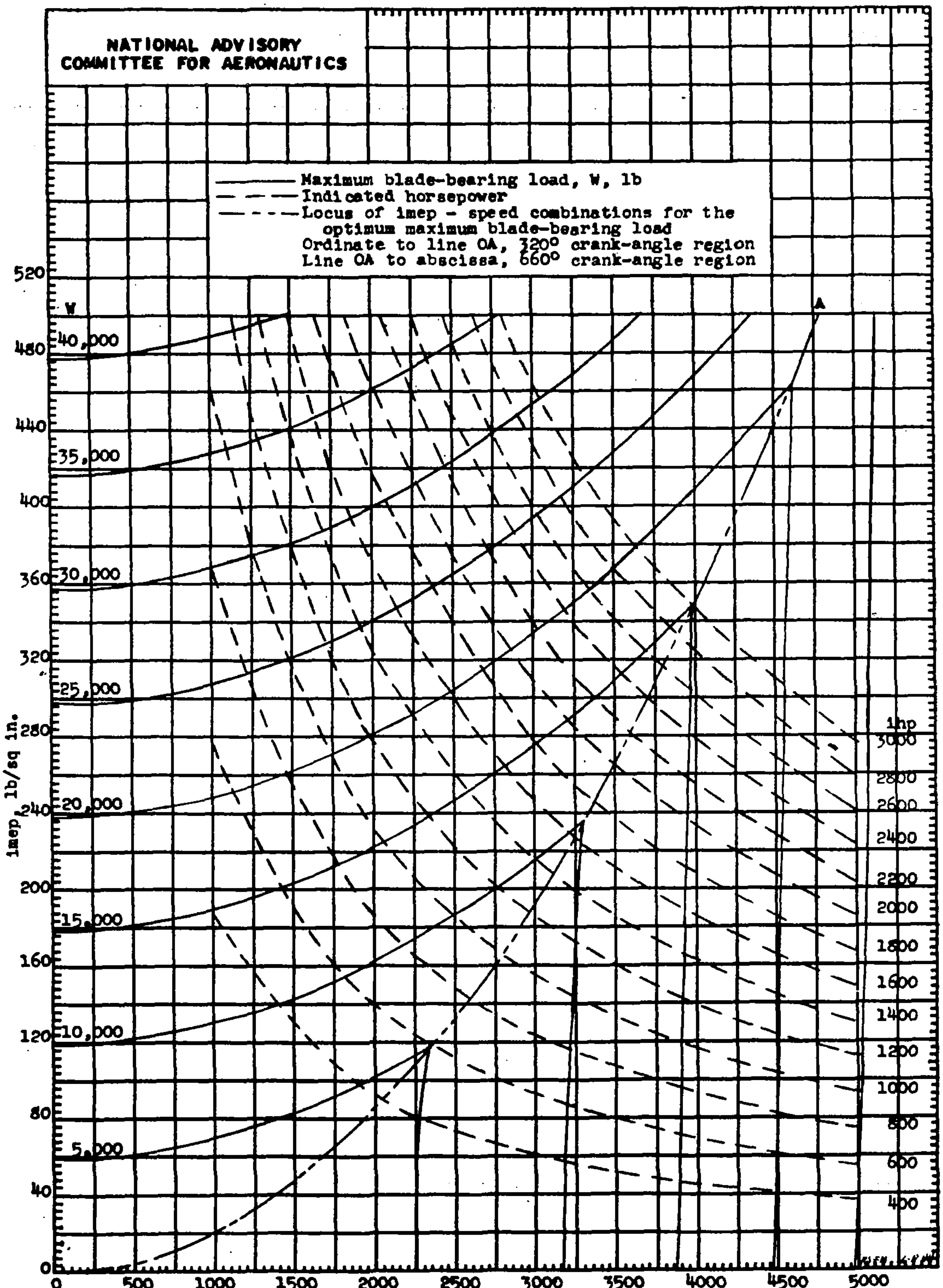
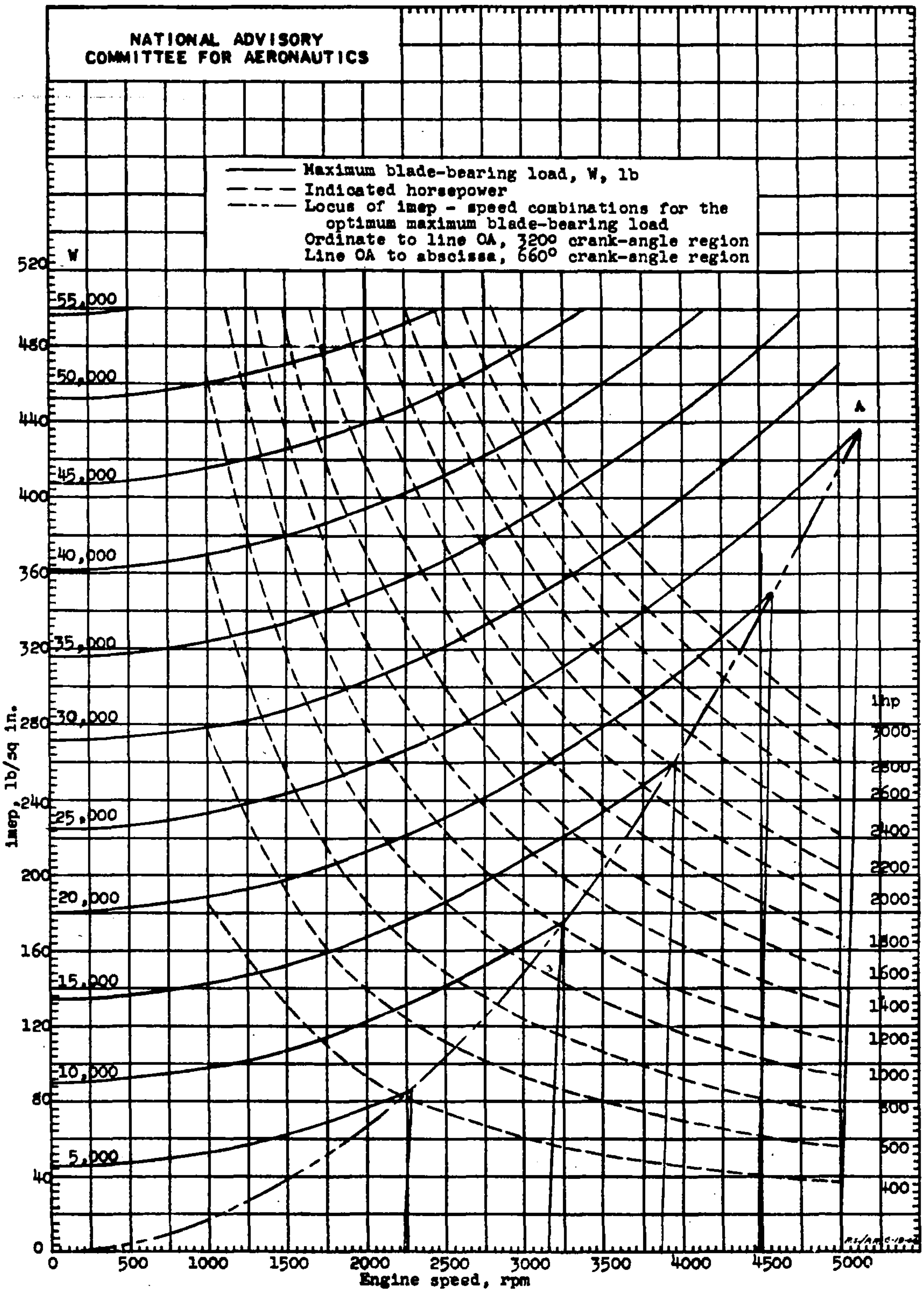
Figure 13. - Polar diagram showing the magnitude of the resultant force on the blade journal of a V-type engine and its direction with respect to the engine axis. Engine speed, 3600 rpm; indicated mean effective pressure, 182 pounds per square inch.

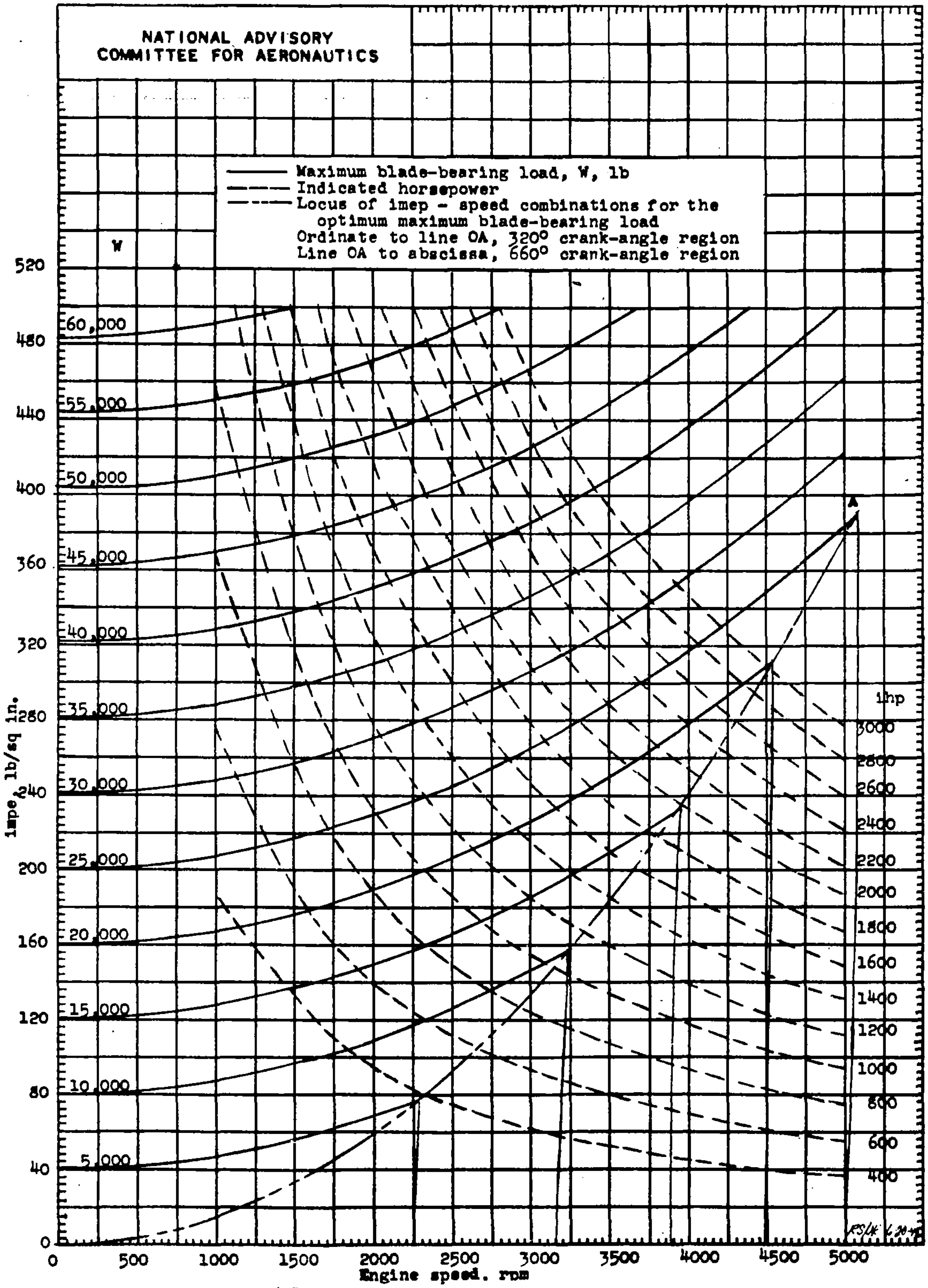


(a) Compression ratio, 5.50.

Figure 14. - Maximum load on blade bearing of a production, V-type engine for all values of indicated mean effective pressure and engine speed at various compression ratios. (Constant maximum-load curves.) Effective bearing area, 3.44 square inches.



(b) Compression ratio, 7.50.
Figure 14. - Continued.



(c) Compression ratio, 8.50.
Figure 14. - Concluded.

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